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The Role of Elastohydrodynamic Lubrication

in Rolling-Contact Fatigue N 64 10 175*

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The five-ball fatigue tester was used to determine the rolling-contact fatigue life of $1/2$ -inch-diameter M-1 steel balls with four lubricants at 300 deg F. Film thickness measurements were made with the rolling-contact disk machine under simulated five-ball test conditions. Under certain conditions, elastohydrodynamic lubrication was found to exist at initial maximum Hertz stress levels up to 800,000 psi. There appears to be a correlation among the following variables: Plastically deformed profile radius of the ball specimen at ambient temperature; lubricant type; and rolling-contact fatigue. No correlation was found between contact temperature obtained with different lubricants and fatigue life.

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Introduction

ROLLING-ELEMENT bearings are used in missile, aircraft, and space applications where operating conditions are so severe that lubrication and operating reliability become serious problems. These operating conditions dictate the need to develop lubricants with better thermal and oxidative stability.

Bearing failure by fatigue is affected by the physical and chemical properties of the lubricant. Knowledge of how these chemical and physical properties affect rolling-contact fatigue would be a useful guide in developing new lubricant formulations.

Differences in the properties of lubricants can affect at least three variables during rolling contact: contact temperature [1 and 2];⁴ contact geometry [3 and 4]; and the minimum separation of the surfaces in contact (lubricant film thickness) [4]. The heat generated and the temperature in the zone of contact of two rolling surfaces are functions of the rheological and thermal properties of the lubricant, so that contact temperatures may vary with different lubricants under dynamic conditions. The generation of heat within the lubricant film induces thermal gradients and thus thermal stresses in the material, which in turn can affect the shearing stresses in the material as calculated from theory [5]. This increase in shearing stresses

may then account for differences in fatigue lives exhibited with different lubricants since life varies inversely with stress to the 9th power.

Changes in contact geometry produce differences in contact stress. Therefore, since deformation of the rolling-element surfaces affects contact geometry, it may play a significant role in the failure of rolling-element mechanisms.

Under dynamic conditions, the contact geometry of two rolling elements can be markedly different from that calculated for static conditions and that measured after deformation and wear. To establish the trends in contact stresses in rolling contact under dynamic conditions and to determine the validity of elastohydrodynamic theory, dynamic profile and film thickness measurements are essential.

In order to understand the role of a lubricant in the rolling-contact phenomenon, it is necessary to measure such characteristics as film thickness, surface deformation and wear, and contact temperature. Therefore, the object of the research described in this paper, which is based on the work reported initially in reference [6], was to investigate with four lubricants the effect of lubricant type or base stock on rolling-contact fatigue life, to determine experimentally the effect of lubricant type and elastohydrodynamic lubrication on film thickness and contact geometry, and to determine experimentally a relation among contact temperature, lubricant type, and fatigue life. All the results for a given lubricant were obtained from the same lubricant batch.

Apparatus

Two types of rigs, the five-ball fatigue tester and the rolling-contact disk machine and X-ray system, were used in this investigation. The five-ball fatigue tester, described initially in reference [7] and shown in Figs. 1(a) and (b), consists essentially of a test specimen pyramided upon four lower support balls,

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⁴ Numbers in brackets designate References at end of paper.

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Nomenclature

a = deformation and wear area from surface trace, sq in. ($a = D + W$)
 B_{10} = ball specimen 10-percent life, millions of stress cycles
 B_{50} = ball specimen 50-percent life, millions of stress cycles
 C = thrust-load capacity, the load at which 90 percent of a group of bearings can endure 1 million inner-race revolutions, or, for ball specimens, 1 million stress cycles, lb ($C = P\sqrt{L}$)
 C_s = system thrust-load capacity of

NASA five-ball fatigue tester, lb
 C_T = thrust-load capacity of full-scale bearings, lb
 D = deformation area from surface trace, sq in.
 H = depth of running track from surface trace, in.
 L = ball specimen or bearing 10-percent life, or life at which 90 percent of a group of specimens remains unfailed, millions of stress cycles or inner-race revolutions
 L_{10} = bearing 10-percent life, millions of inner-race revolutions

L_{50} = bearing 50-percent life, millions of inner-race revolutions
 P = bearing or test-system thrust load, lb
 R = radius of ball, in.
 R_p = effective radius of ball profile after plastic deformation and wear, in.
 W = wear area from surface trace, sq in.
 β = contact angle, deg
 ω = angular velocity of drive shaft, rpm
 ω_s = angular velocity of spin of ball specimen relative to support balls, rpm

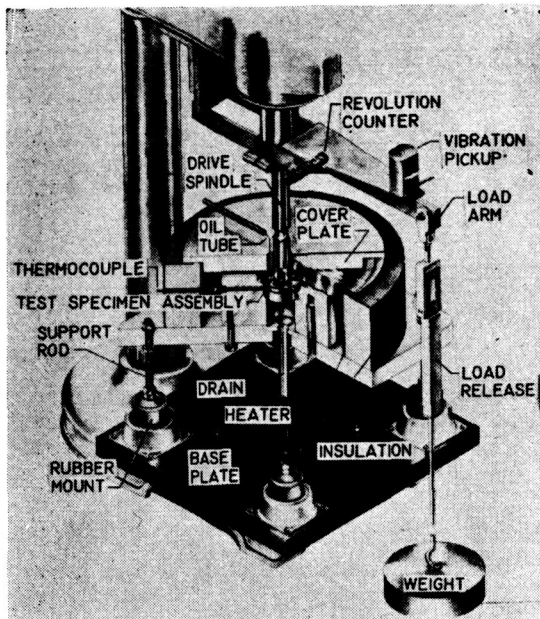


Fig. 1(a) Five-ball fatigue tester

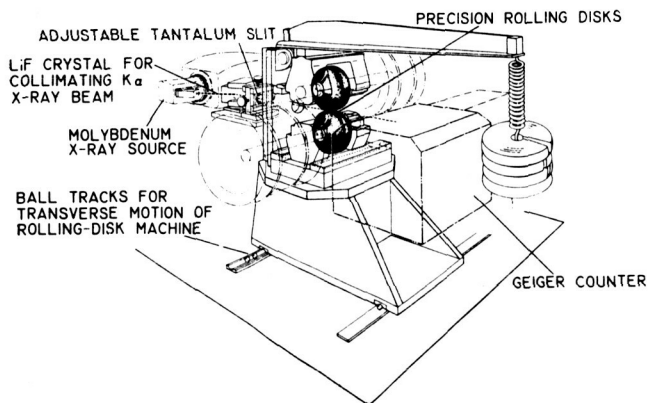


Fig. 2 Precision rolling-disk machine and X-ray system

positioned by a separator, and free to rotate in an angular contact raceway, Fig. 1(b). Specimen loading and drive are supplied through a vertical shaft. For every revolution of the drive shaft, the test specimen receives three stress cycles. By changing the pitch diameter of the support balls, the contact angle β can be controlled. Instrumentation provides for automatic failure detection and shutdown and thus makes long-term unmonitored tests possible.

The five-ball fatigue tester was modified to measure the temperature near the contact area of a modified test specimen during operation. Fig. 1(c) shows the test specimen and mounting assembly, which is inserted into the drive spindle of the five-ball rig [see Fig. 1(a)]. Each test specimen had a thermocouple attached with the tip at one edge of the running track. An axial hole was drilled through the drive spindle to accept the thermocouple wire. The thermocouple emf was taken out through a slip-ring brush assembly mounted at the top of the drive spindle.

The rolling-contact disk machine is shown in Fig. 2. This machine, described in detail in references [3 and 4], was used to measure lubricant film thickness and contact deformation under dynamic conditions. Essentially, this method of measuring film thickness consists of directing a monochromatic, collimated, square beam of high-energy X rays between two rolling-disk surfaces. The amount of radiation passing between the disk parallel to the flat contact regions is related to the thickness of the lubri-

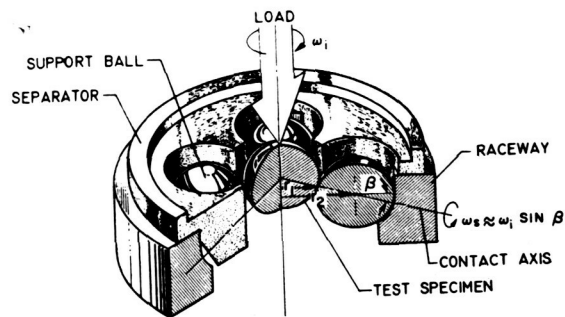


Fig. 1(b) Schematic, five-ball tester

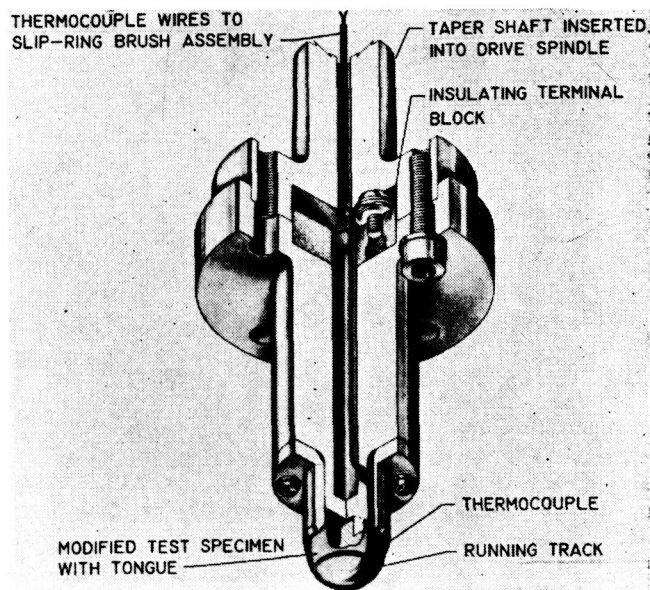


Fig. 1(c) Operating-temperature measuring device

Fig. 1 Test apparatus

cant film separating the surfaces. A particular wavelength X-ray was selected that penetrated lubricants quite readily but did not penetrate steel significantly.

The X-ray system was fitted with apertures to limit the width of the X-ray beam to 10 mils in order to obtain sufficient delineation of the contact-deformation profiles for the highly crowned disks on which the contact width was only 20.5 mils at 800,000-psi maximum Hertz stress (assuming no plastic deformation).

Materials and Procedure

Four lubricants of practical interest were selected for study in this program. They can be classified into two basic groups: ester and mineral oil. A summary of the properties of these lubricants can be found in Table 1. Fig. 3 is an ASTM standard viscosity-temperature chart for these lubricants.

Fatigue tests were conducted with each of these lubricants in the five-ball fatigue tester with $\frac{1}{2}$ -inch M-1 steel balls of a Rockwell C-63 hardness. All the balls for each test came from the same heat of material. Standard test conditions for these tests were a temperature of 300 deg F, a drive shaft speed of 10,000 rpm, a 30-deg contact angle, and a ball loading that produced a maximum Hertz stress of 800,000 psi.

To determine the variation in contact temperature with lubricant type, temperature measurements were taken at the edge of the running track under different operating conditions with each of the four lubricants in the modified five-ball fatigue tester. Temperature measurements were taken for each lubricant at an initial maximum Hertz stress of 725,000 psi, contact angles of 10

Table 1 Test lubricant properties

Lubricant type	Lubricant designation	Base stock	Additive content		Neutralization number (before test)	Viscosity, cs (Fig. 3)		
			Type	Vol, %		100° F	210° F	300° F
Ester	NA-XL-3	Complex ester of octyl alcohol, adipic acid, and polyethylene glycol Di-2-ethylhexyl sebacate	(1) Oxidation inhibitor	0.5	0.18	35.30	7.14	3.40
	(1) Oxidation inhibitor (2) Extreme pressure additive (3) Antifoam agent		0.5	0.25	13.29	3.51	1.86	
			5.0					
				0.0005				
Mineral oil	NA-XL-4	% Carbon paraffinic-63 % Carbon naphthenic-32 % Carbon aromatic-5	(1) Extreme pressure additive	2.0	0.09	126.60	11.72	4.30
	NA-XL-7	% Carbon paraffinic-66 % Carbon naphthenic-33 % Carbon aromatic-1	No additives	...	0.10	238.70	20.68	7.00

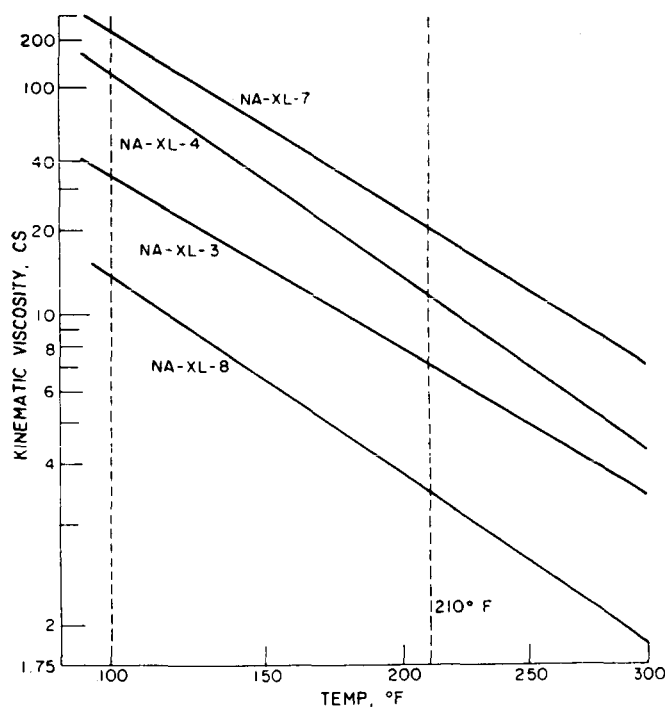


Fig. 3 Viscosity of experimental lubricants as a function of temperature

and 30 deg, and ambient temperature. Measurements were also made at an initial maximum Hertz stress of 800,000 psi, a 30-deg contact angle, and two race temperatures, ambient and 300 deg F.

Deformation and wear data were obtained for M-50 steel balls of hardness Rockwell C-62 tested at an initial maximum Hertz stress of 800,000 psi, a contact angle of 10 deg, no heat added, and 30,000 stress cycles in the five-ball fatigue tester with each of the four test lubricants. For each lubricant, eight balls were run. Six profile traces of each ball were made in a contour tracer at different locations around the ball, perpendicular to the running track. Wear and deformation areas were measured from these transverse profile traces of the running track [6 and 7].

In order to measure film thicknesses and determine trends in contact geometry under dynamic conditions simulating those in the five-ball fatigue tester, the precision rolling-contact disk

machine was used. Rolling disks of hardened and stabilized SAE 52100 steel were fabricated for this machine with a transverse radius of 0.44 inch machined on the 2.88-in-diameter disk surfaces. Film thickness and contact profiles were measured with the four test lubricants over the following range of conditions: initial maximum Hertz stresses, 0 to 800,000 psi; disk surface temperatures, ambient to 300 deg F; and disk tangential speeds, 850 to 2600 fpm.

Results and Discussion

Fatigue Life. Rolling-contact fatigue data are plotted on Weibull plots. A Weibull plot is a plot of the log-log of the reciprocal of the probability of survival against the log of stress cycles to failure. For convenience, the ordinate is graduated in statistical percent of specimens failed.

For aircraft and spacecraft applications, high reliability is of paramount importance. Therefore, for comparison purposes, the significant life on the Weibull plot is the 10-percent life. The 10-percent life is the number of stress cycles within which $1/10$ of the specimens can be expected to fail. (This 10-percent life is equivalent to a 90-percent probability of survival.) The failure index that is given with each plot indicates the number of specimens failed out of those tested.

Weibull plots for the four lubricants tested with M-1 steel balls are shown in Fig. 4. These data are tabulated in Table 2. The Material Laboratory of the New York Naval Shipyard tested these same lubricants (from the same lubricant batch) with 7208-size thrust ball bearings at 300 deg F, a 40-deg contact angle, and a 4600-pound thrust load producing maximum Hertz stresses of 349,000 and 322,000 psi on the inner and outer races, respectively [8]. These data are plotted in Fig. 5 using the methods of reference [9] and are summarized in Table 2.

An important criterion of bearing operation is its load-carrying capacity, usually termed simply capacity. This is the load that can be carried for 1 million stress cycles or inner-race revolutions. The load-carrying capacity with each lubricant tested may be calculated from the fatigue life results summarized in Table 2 where

$$C = P \sqrt[3]{L}$$

The load capacity for each lubricant tested is given in Table 3. It should be noted that the fatigue life and the load capacity

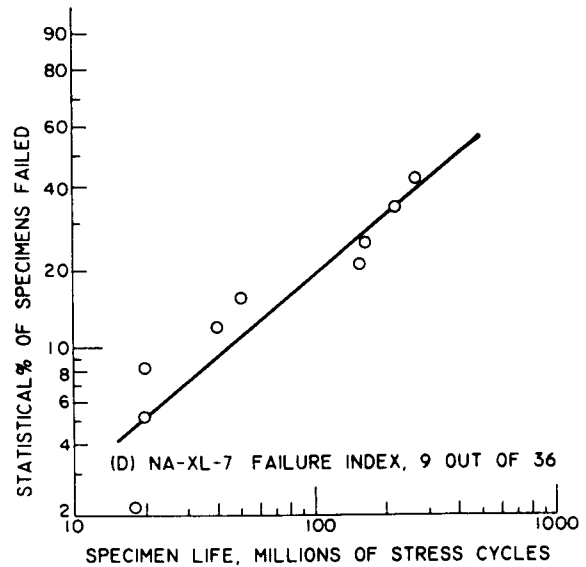
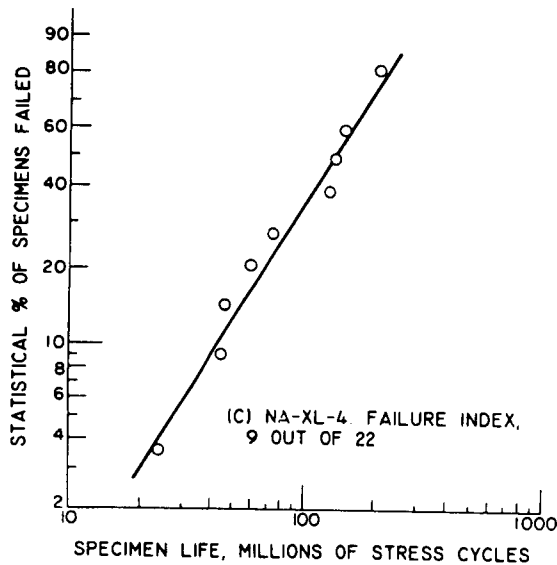
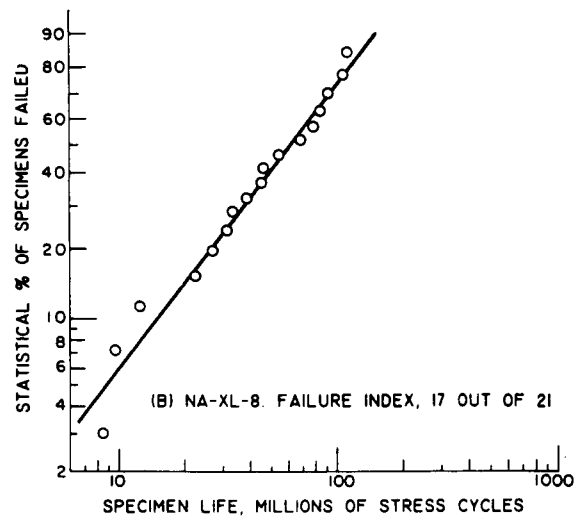
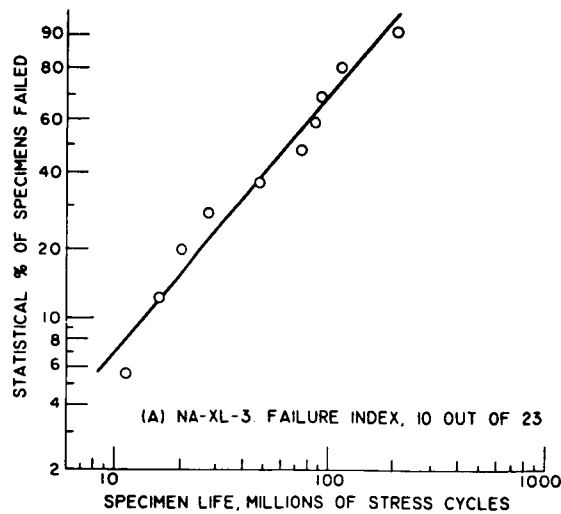


Fig. 4 Rolling-contact fatigue life of AISI M-1 steel balls with various lubricants [6]. Race temperature, 300 deg F; maximum Hertz stress, 800,000 psi; five-ball fatigue tester.

Table 2 Test lubricant fatigue results

Lubricant type	Lubricant designation	Five-ball fatigue tester; 300° F; maximum Hertz stress, 800,000 psi				7208-size bearing; 300° F; thrust load, 4600 lb			
		Stress cycles		Ratio of B_{10} life to B_{50} life of NA-XL-7	B_{10} life confidence number ^a	Inner-race revolutions		Ratio of L_{10} life to L_{50} life of NA-XL-7	L_{10} life confidence number ^a
		B_{10}	B_{50}			L_{10}	L_{50}		
Ester	NA-XL-3	13.7×10^6	6.50×10^6	0.31	78%	1.1×10^6	4.4×10^6	0.33	87%
	NA-XL-8	14.9	60.0	0.34	81	1.4 ^b	8.8 ^b	0.49 ^b	86 ^b
Mineral oil	NA-XL-4	42.5	132.0	0.96	53	2.5	17.0	0.76	58
	NA-XL-7	44.5	395.0	1.00	...	3.3	21.0	1.00	...

^a Percent of time that 10-percent life obtained with each lubricant will have the same relation to the 10-percent life of NA-XL-7.

^b Not from same lubricant batch.

Table 3 Load capacity of test lubricants

Lubricant type	Lubricant designation	Five-ball fatigue tester, 300° F		7208-size bearing, 300° F	
		C_s , lb	Ratio of C_s to C_s of NA-XL-7	C_T , lb	Ratio of C_T to C_T of NA-XL-7
Ester	NA-XL-3	818	0.68	4750	0.69
	NA-XL-8	842	0.70	5150	0.75
Mineral oil	NA-XL-4	1190	0.98	6240	0.91
	NA-XL-7	1210	1.00	6850	1.00

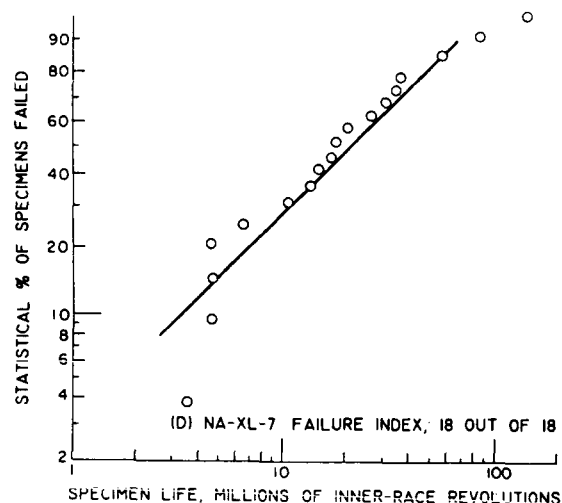
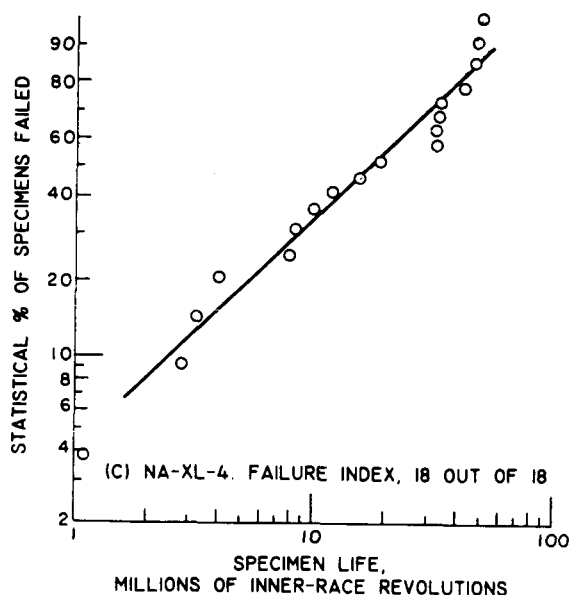
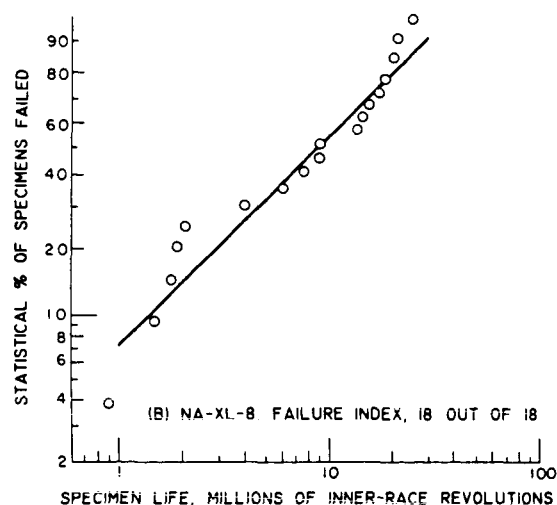
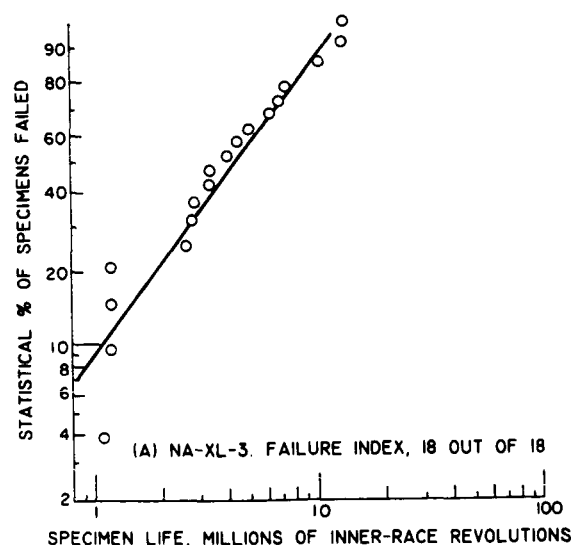


Fig. 5 Rolling-contact fatigue life of 7208-size angular ball bearings with various lubricants [8]. Temperature, 300 deg F; speed, 3450 rpm; axial load, 4600 lb.

data for each of these lubricants are compared relative to that obtained for lubricant NA-XL-7.

A comparison is made in Fig. 6 between the relative capacity of the lubricants run in the five-ball rig and the full-scale bearings that indicates a correlation between this rig and the bearings.

Film Thickness Measurements. In order to understand the role of elastohydrodynamics in the rolling-contact phenomenon, the lubricant film thickness and the shape of the deformed contact regions were measured in the rolling-contact disk machine during operation. These measurements were taken at rolling velocities, stresses, and temperatures that simulated those in the five-ball fatigue tester.

Lubricant film thickness at the center of the contact zone is plotted for lubricant NA-XL-7 in Fig. 7. Film thicknesses are shown as a function of the atmospheric-pressure viscosity at the measured disk temperature for a constant disk tangential surface speed of 1350 fpm. Two curves representing Archard's elastohydrodynamic theory for "point contacts" [10] are plotted in Fig. 7 for comparison with the experimental curves taken under the same conditions.

The lubricant film was not measurable by the X-ray technique at viscosities below a certain critical viscosity (about 50 cs at 800,000-psi maximum Hertz stress and about 10 cs at 320,000-psi maximum Hertz stress). No significant X-ray count above the background level could be detected through the center of the

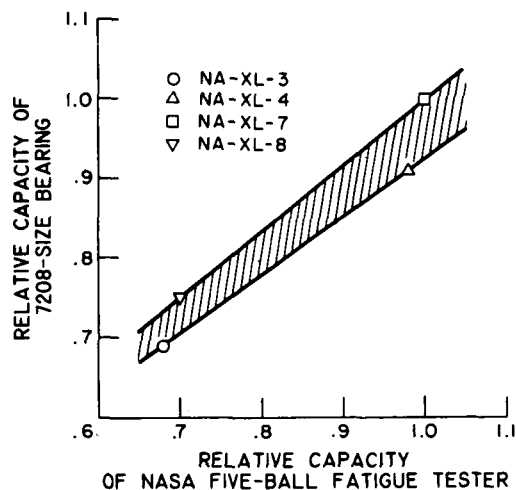


Fig. 6 Relative load-carrying capacity of a 7208-size bearing as a function of relative capacity of NASA five-ball fatigue tester for various lubricants at 300 deg F

contact under these conditions; that is, the contact behaved almost as if the disks were loaded together statically. The absence of a measurable oil-film thickness by the X-ray method, however, does not necessarily indicate the presence of boundary lubrication. Since the length of the high-pressure contact region through

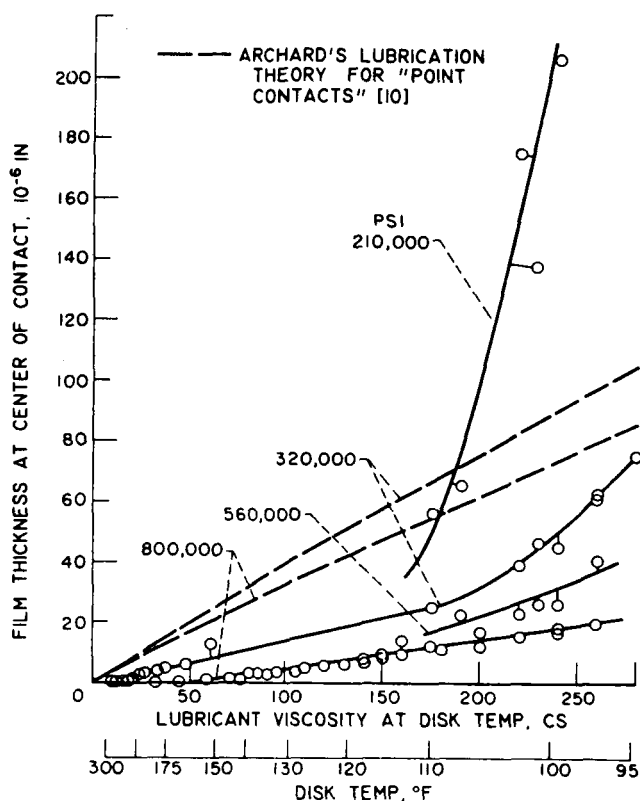


Fig. 7 Effect of maximum Hertz stress and lubricant viscosity on film thickness for NA-XL-7; 1350 fpm rolling speed

which the X-ray beam was transmitted in these experiments was about 0.1 in., considerable loss of beam intensity could be accounted for by absorption by the oil and by nonplanarity of the contact region owing to uneven deformation.

Contact-deformation profiles obtained with the four test lubricants at a maximum Hertz stress of 800,000 psi, a disk tangential speed of 850 fpm, and disk temperatures of 112 and 300 deg F are compared in Figs. 8(a) and (b). It should be noted that these profiles are highly magnified in the y -direction relative to the x -direction.

Comparison of the deformed profile data of Fig. 8(a) and the fatigue life data obtained at room temperature in the NASA fatigue spin rig [6] indicates that better fatigue life is obtained with narrower contact profiles. The data are hardly conclusive, especially in view of the fact that the contact profiles in the rolling direction were not determined. More work needs to be done in this area.

At the high temperatures [Fig. 8(b)] direct comparison of the dynamic contact deformation profiles is difficult. Although the NA-XL-8 produced the narrowest contact, the order of the other three lubricants is not clear.

Typical contact profiles and film thicknesses with NA-XL-7 at initial maximum Hertz stresses up to 560,000 psi and at an average disk temperature of 100 deg F are shown in Fig. 9(a). Measurement of the profiles of the contact region under no load after running indicated no plastic deformation up to maximum Hertz stresses of 600,000 to 650,000 psi. (This was also noted with the three other test lubricants.) However, at maximum Hertz stresses beyond these values, plastic deformation was detected. Fig. 9(b) shows the dynamic profile at maximum Hertz stress levels up to 800,000 psi, where plastic as well as elastic deformation occurred. These data plotted in Fig. 9 indicate that film thickness decreases rapidly with increasing load at contact stresses below which plastic deformation occurs. At higher stress levels small decreases in film thickness occur with large increases in load.

The data for these four lubricants indicate a thinning of the lubricant film at the edges of the contact zone for "point contact."

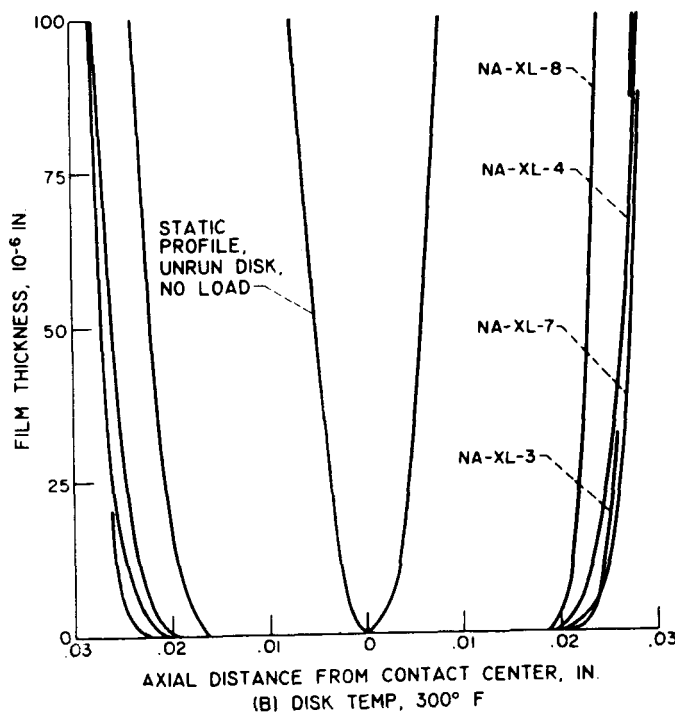
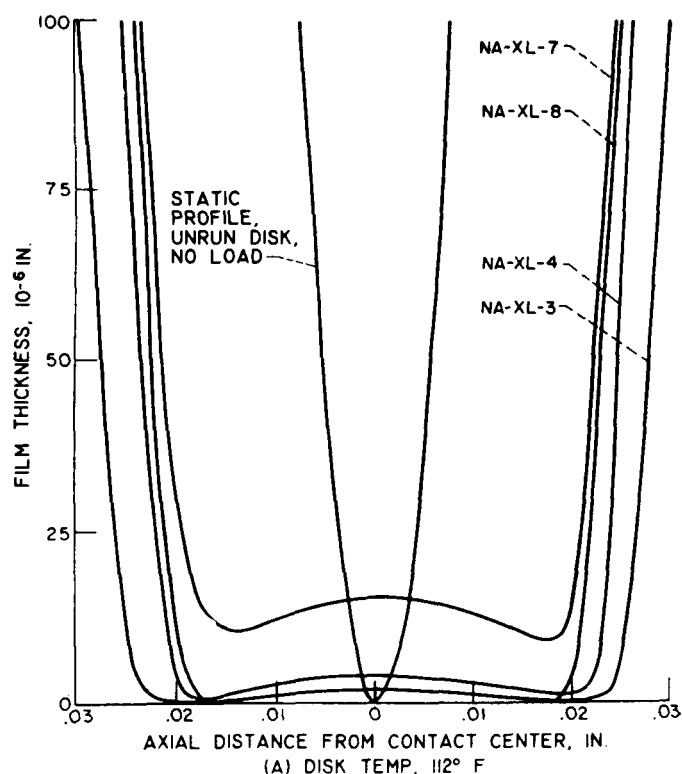


Fig. 8 Comparison of contact profiles for different lubricants. Maximum Hertz stress, 800,000 psi; disk tangential speed at center of contact, 850 fpm.

Two-dimensional cylinder theory [11 and 12] has predicted the same phenomena but at the trailing edge of the contact zone. This has been verified in reference [13].

Rolling speed changed the oil-film thickness more significantly than the contact deformation profile as illustrated in Fig. 10.

The results reported herein indicate that hydrodynamic conditions can prevail at rolling contacts under stresses and speeds simulating those in a five-ball fatigue tester even when the ball specimens are deformed plastically. The minimum film thickness evidently becomes very small at high temperatures and with low

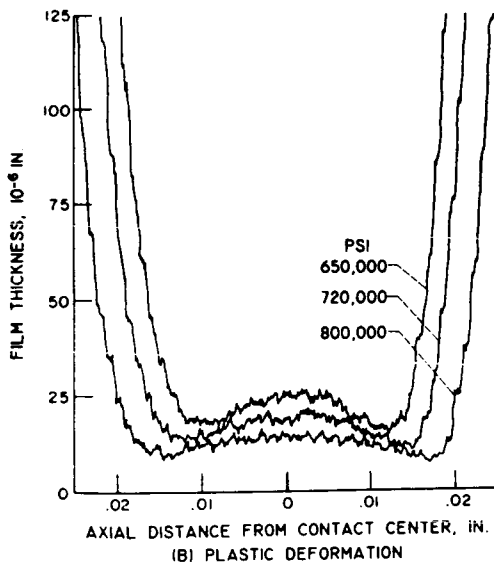
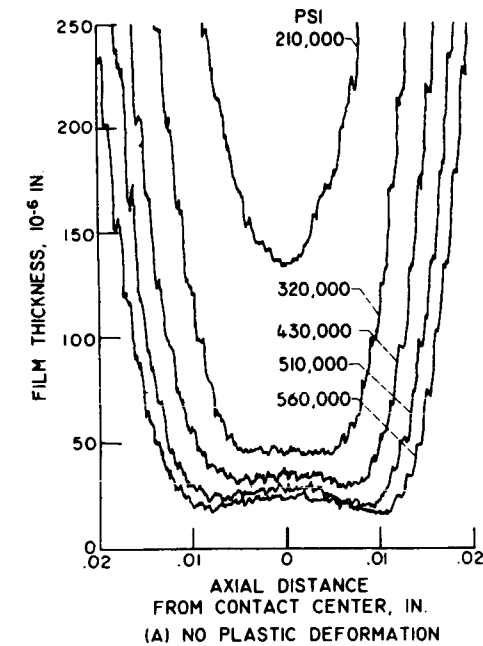


Fig. 9 Contact profiles and film thickness as a function of maximum Hertz stress with NA-XL-7. Disk tangential speed at center of contact, 1350 fpm; disk temperature, 100 deg F.

viscosity lubricants. However, the significant differences in the shape of the deformed regions detected with different lubricants indicate that appreciable hydrodynamic pressures still exist under these low-viscosity, high-temperature conditions. These differences in deformed profile caused by the lubricant may have important effects on contact stresses and bearing life.

Deformation and Wear. As previously indicated, the lubricant alters the contact stress and contact geometry of rolling elements from those calculated for static conditions. The alteration produced on the rolling-contact surfaces takes three basic forms: (a) elastic deformation; (b) plastic deformation; and (c) wear. The latter two forms result in permanent alteration of the ball surface contour that can be measured after testing. Fig. 11 is a schematic diagram of the transverse section of a ball surface showing this permanent alteration.

Deformation and wear data with each of the four test lubricants were obtained for M-50 balls under the conditions previously described. Average values for the deformation and wear areas are given in Table 4. By the use of trigonometric relations, an effective ball radius R_p at the point of contact can be calculated [6] in terms of a , H , and R :

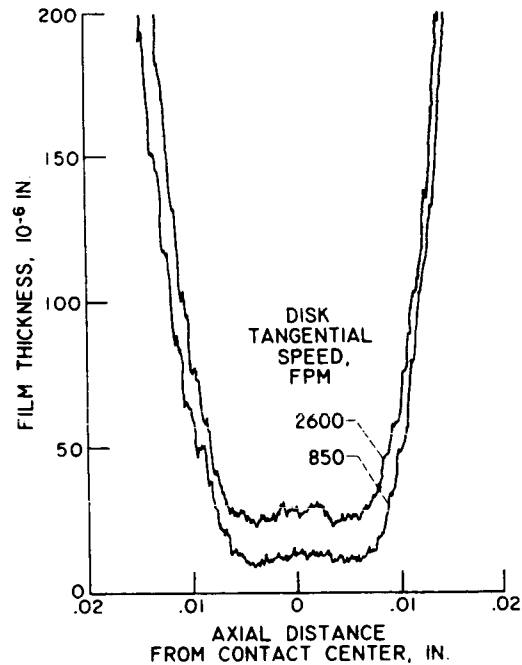


Fig. 10 Effect of speed on contact profiles with NA-XL-7. Maximum Hertz stress, 320,000 psi; disk temperature, 121 deg F.

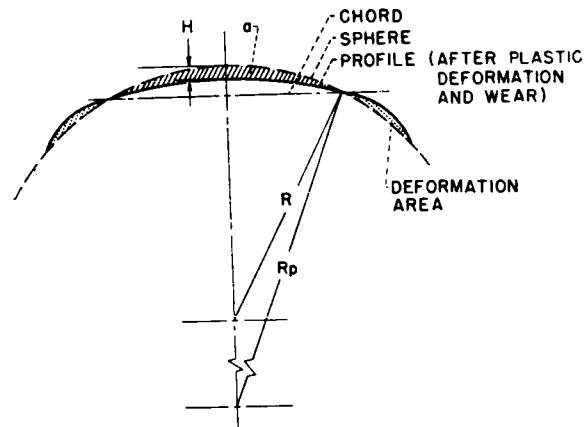


Fig. 11 Surface profile after deformation and wear of cross section of stressed ball track. (Not to scale.)

$$R_p = \frac{(a/H)^2}{2[R - \sqrt{R^2 - (a/H)^2} - H]}$$

On the basis of this equation for R_p , an effective maximum Hertz stress for 30,000 stress cycles of operation was calculated for each lubricant. These values are also given in Table 4. Based on previous experience, the value of R_p obtained after 30,000 stress cycles approximates the value that would be obtained after an indefinite number of stress cycles.

The differences in the recalculated static Hertz stress for specimens run with each lubricant does not completely account for the observed differences in life. However, a plot (Fig. 12) of the fatigue life for each lubricant tested in the five-ball fatigue tester as a function of the deformed profile radius indicates that increasing life is obtained with increased profile radius (deformation). While these data are limited, they do suggest that the hydrodynamic forces in the lubricant film do distort the stress pattern calculated from static force conditions and can be great enough to account for the observed differences in fatigue life.

Temperature Studies. Temperature data were obtained in a modified five-ball fatigue tester for the four test lubricants and are given in Table 5. These temperatures are a better approximation of the actual temperature in the contact zone of a ball specimen than temperatures measured at the outside diameter of the race.

Table 4 Deformation and wear and their effect on maximum Hertz stress for M-50 steel balls tested in five-ball fatigue tester at 800,000-psi initial maximum Hertz stress, 10 deg contact angle, 30,000 stress cycles, and room temperature

Lubricant type	Lubricant designation	Deformation area from surface trace, D, sq in.	Wear area from surface trace, W, sq in.	Calculated profile radius, R_p	Effective maximum Hertz stress, psi	
					No deformation and wear of support balls assumed	Deformation and wear of support balls assumed equal to test ball
Ester	NA-XL-3	4.17×10^{-7}	3.50×10^{-7}	0.329	7.65×10^6	7.30×10^6
	NA-XL-8	3.83	4.17	0.322	7.67	7.34
Mineral oil	NA-XL-4	5.17	4.33	0.346	7.59	7.18
	NA-XL-7	4.33	4.00	0.335	7.63	7.25

Table 5 Temperature at edge of contact zone, modified five-ball fatigue tester, 10,000 rpm

Lubricant type	Lubricant designation	Temperature, deg F (no heat added)		
		Maximum Hertz stress, 725,000 psi		Maximum Hertz stress, 800,000 psi
		$\beta = 10$ deg	$\beta = 30$ deg	$\beta = 30$ deg
Ester	NA-XL-3	117	141	161
	NA-XL-8	118	133	156
Mineral oil	NA-XL-4	141	187	213
	NA-XL-7	146	189	218

It has been previously suggested that these thermal stresses affect fatigue life to the extent that lubricants yielding the lowest temperature in the contact zone will give the longest fatigue life in rolling contact [1]. However, when the temperature measurements (Table 5) are compared with rolling-contact fatigue life results (Table 2), it appears that the lubricants giving the longest lives yield the highest temperatures. These temperatures were measured with the system running without heat addition. At an outer-race temperature of 300 deg F, little difference was found in the temperatures at the edge of the contact zone despite wide variations in fatigue life with different lubricants. These results tend to indicate that thermal stresses due to temperature gradients in the contact zone of two rolling bodies may be only of secondary importance in resolving differences in fatigue life.

Summary

The five-ball fatigue tester was used to determine the rolling-contact fatigue life of 1/2-inch M-1 steel balls with four experimental lubricants. These lubricants were classed as esters and mineral oils. Tests were run at an initial maximum Hertz stress of 800,000 psi and a 30-deg contact angle at 10,000 rpm. Profile and film thickness measurements were taken under dynamic conditions in the rolling-contact disk machine. Plastic deformation and wear of the specimens in rolling contact were also studied. Temperature measurements were taken at the edge of the contact area. The following results of these investigations were obtained:

1 Hydrodynamic conditions can prevail at rolling contacts simulating those in the five-ball fatigue tester even when the ball specimens are deformed plastically, as in the experiments described in this paper.

2 The contact geometry is affected by the rheological properties of these lubricants. A correlation is suggested between the plastically deformed profile radius (at ambient temperature after the first few cycles of operation), lubricant type, and rolling-contact fatigue life.

3 No correlation was found between fatigue life and contact temperature induced by different lubricants. This lack of correlation suggests that thermal stresses induced by temperature gradients may be only of secondary importance in accounting for differences in fatigue life with various lubricants.

4 Relative fatigue life and capacity results obtained in the five-ball fatigue tester compared favorably with those obtained with 7208-size bearings tested with the same lubricants.

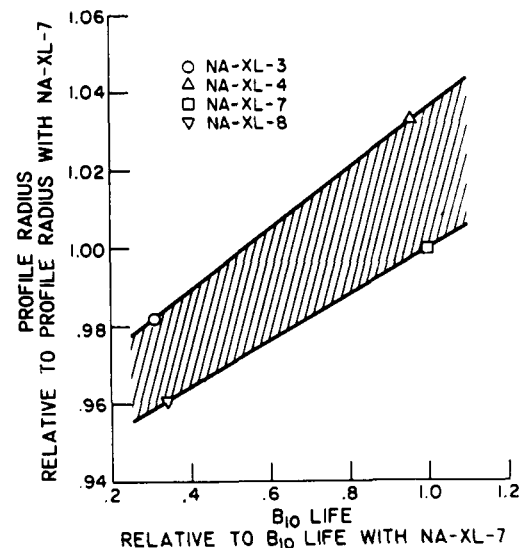


Fig. 12 Profile radius measured at a 10 deg contact angle, no heat added, plotted against B_{10} life, 30 deg contact angle, 300 deg F outer-race temperature. Initial maximum Hertz stress, 800,000 psi, five-ball fatigue tester.

Acknowledgments

The authors would like to acknowledge the invaluable help of the personnel from the NASA Lewis Research Center and the Battelle Memorial Institute who assisted in this research. We would also like to thank the Aeronautical Systems Division of the United States Air Force Systems Command for supplying the experimental lubricants.

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DISCUSSION

R. A. Burton⁵

The authors are to be congratulated on this useful and important paper. It provides an excellent example of how two laboratories, each with special skills and apparatus, can combine their efforts with exceptional results. In many ways this paper may be looked upon as extending earlier work by providing a mechanical correlate to the observed differences in fatigue life shown by different oils. The authors' arguments and graphs show significant differences in contact stresses and contact profiles for the different oils, and these appear to bear a regular relationship to the lives measured in fatigue tests. Though changes of maximum Hertz stress due to the changes in contact profile are small, it is possible that they may nevertheless explain the differences observed, since earlier work has established that fatigue life bears an inverse relationship to the ninth or tenth power of stress. (This same law has a second manifestation as shown in Fig. 6; even though fatigue life under constant load ranged fourfold, the load-carrying capacity for a given life only ranged down about 25 percent from the highest to the lowest.) Despite the extreme effect of a tenth power law, the information presented does not suggest sufficiently large differences in Hertz stress to account for the differences in lives. This does not mean that the observed changes of curvature may not be related in additional ways to fatigue life, however.

If one looks at the deeper subsurface stresses (a sizable fraction of the contact diameter below the surface), one finds that these stresses are not affected greatly by the details of the pressure distribution on the surface, but by the general magnitude and radial extent of the force distribution. Thus, if deep subsurface stresses dominate failure, the abovenoted gross differences in load distribution would have been required. Since my interpretation of the evidence suggests that this does not occur in sufficient magnitude, it would follow that these deeper stresses must not be the important ones.

On the other hand, surface stresses are affected very strongly by the details of the pressure distribution. When one looks at the overall characteristics of the pressure profile and does not see sufficient justification to believe that the deeper stresses have changed, it is reasonable then to suggest that it is not these stresses but those nearer the surface that dominate in fatigue failure. This does not presuppose that failures must occur always directly on the surface, but that the probability distribution of the failures is biased toward a maximum at or near the surface of the specimen. Furthermore, hypotheses of a deeper maximum-probability zone have largely been based upon shakily established applicability of evidence from bulk fatigue tests.

One cannot show definitely that there are necessary and sufficient conditions to prejudice failures toward the surface, but Fig. 13 is offered here to illustrate a unique aspect of the variation of the octahedral stress along a diameter of a circular contact. This shows that the stress passes through three peaks for every pass the ball makes across a given point on the surface. Furthermore, the magnitude of this stress is comparable to the magnitude of deeper internal stresses, though it is slightly smaller.

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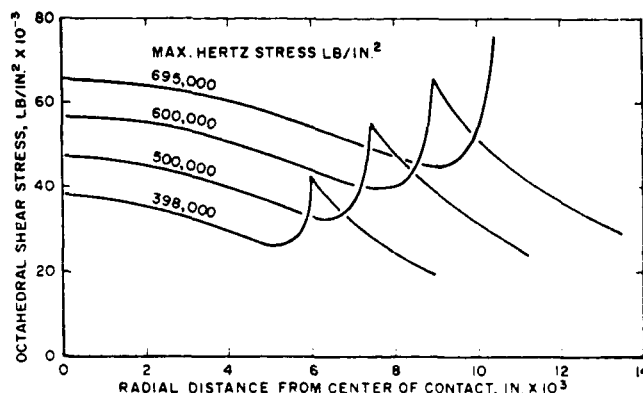


Fig. 13

One must note also the secondary peaks near the edge of the contact, which are in fact caused by the discontinuity of loading at the edge of a true Hertzian contact. It would not be at all surprising to see this feature altered considerably by lubricant effects which alter in a small way the stress distribution at entry and exit of the contact. Furthermore, it would not be surprising for such effects to be directly correlated with the observed curvature effects reported in the paper. The foregoing observations are not intended as counter arguments to the paper, but as an optimistic suggestion toward the next step in attack on the problem.

The authors have shown that in plausible stress ranges, lubrication is hydrodynamic. They have shown that there are observable differences in the behavior of the surface deformations for different oils. They have shown that this behavior correlates with fatigue life. It appears that the next step would be to look again at the theoretical problem of hydroelastic lubrication and to study not only the overall force picture, but stresses in the contact periphery. It may be that observation of these can lead to a clearer understanding of the mechanical role of different lubricants and their effects on fatigue life. If it does become possible to realize the objective of predicting lubricant effects upon stresses, then it will be doubly important to look again at the materials problem and find out just what aspect of the time and spatial variations of stress actually determines the initiation and subsequent growth of cracks.

It is hoped that detailed studies of the type reported will be continued. In any event the evidence provided in this paper will undoubtedly prove repeatedly useful in future studies.

Jerrold W. Kannel⁶

The paper shows that the effect of lubrication on the life of rolling bearings may be qualitatively predictable from knowledge of how lubrication alters the bearing stresses. The authors attempted to infer the effect of bearing stresses on fatigue from a knowledge of the elastic and plastic deformations of the surfaces. However, as was mentioned in the paper, the results could not be considered conclusive since the only deformations available for analysis were those appearing in axial profiles. From theoretical studies, it appears that the deformations that might yield a useful analysis of stresses are the deformations that might be observed in the circumferential profiles of the bearings. For example, elastohydrodynamic theory predicts a sharp "bump" and perhaps a severe pressure concentration on the rolling surfaces near the trailing edge of the contact region. The existence of such deformation and pressure could affect bearing performance greatly.

The rolling-disk machine used in finding the profiles in this paper has been modified recently so that circumferential profiles can be observed. One typical circumferential profile is presented in Fig. 14, along with the corresponding profile under dry contact. The related axial profile, similar to those presented by

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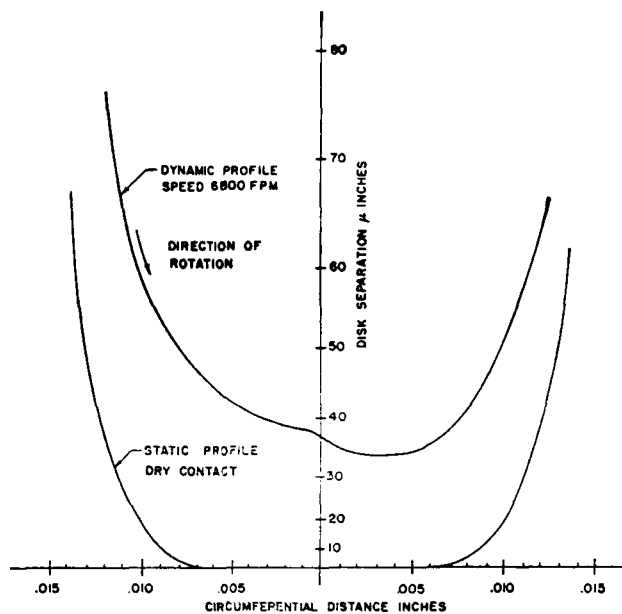


Fig. 14 Circumferential profile deformation

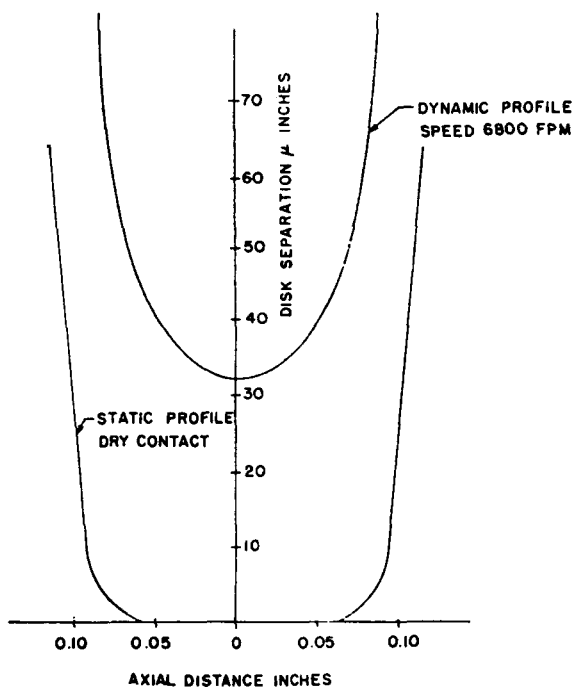


Fig. 15 Axial profile deformation

Mr. Zaretsky, et al., is given in Fig. 15. It can be observed that the influence of the lubricant on the circumferential profile is quite noticeable. Unlike the axial profile, the circumferential profile is not symmetric about its center; the exit radius is notably smaller than the inlet radius. The film thickness is not constant, and at its minimum point which occurs near the trailing edge of the contact region there is an apparently elongated bump. This bump could be related to the severe pressure concentration predicted by the theories.⁷ The profiles were found to depend on temperature, pressure, and rolling speed and should be fairly indicative of the conditions of stress. It would be very interesting, of course, to take both types of profiles and to attempt to infer a three-dimensional map of the deformed

⁷ Since the disks used in this study contained a slight crown, they developed an elliptically shaped contact region as opposed to the rectangular regions assumed by theorists. Thus no simple correlation exists between theory and the experimental data.

surfaces and to compare these with bearing fatigue. To date, no circumferential profiles have been obtained under conditions like those used in obtaining any known fatigue data.

Since the primary goal of any lubrication-bearing stress study is to attempt to predict actual bearing performance, it would be interesting to study fatigue and profile deformations under actual bearing stress conditions. However, the correlation of bearing performances with the accelerated fatigue data given in the paper represents a good beginning point for this type of study and demonstrates the feasibility of the approach.

Bruce W. Kelley⁸

I have watched with interest the attempts to find the cause of the apparent influence of lubricants on surface fatigue of bearings. The authors have made progress in this attempt and the discovery of the phenomena may have a real effect on our understanding of this complex field.

I am not yet convinced of the importance of elastohydrodynamic lubrication in rolling contact as far as its effect on load distribution is concerned—at least at loads great enough to create failure of conventional bearing or hardened gear materials. It is known, however, that frictional characteristics of different lubricants in the thin film or partial hydrodynamic state are markedly different and it may be more impressive to look in this direction. Even though the amounts of sliding within the contact area are small (most of which might be termed "impending slippage") the tractive efforts that can develop are very much a function of the oil. It seems that these tractive efforts may have a far greater effect on contact stresses than the rather minor redistribution of pressures and stresses caused by hydrodynamic effects.

The effect of temperatures created in the contact area encompasses two areas of interest. The first which is discussed in the paper is the existence of thermal stresses. In the case of significant sliding, such as occurs on gears, calculations have shown that within the first 0.005-in. depth from the surface significant thermal stresses can be developed because of the high temperature gradient. From this and the well-defined tractive effort of frictional forces, hardened gear teeth show pitting which appears to originate from the surface rather than below the surface as occurs in well controlled, pure rolling contact. In addition, the surface capacity under such conditions is considerably lower than with pure rolling contact.

The second area which is influenced by contact temperatures is that of material strength and the ease of metallurgical structural change under stress. Since one is hard pressed to find a sizable change in shear stresses caused by thermal stresses which are largely compressive, this latter point may be of more importance.

Clues to the importance of the foregoing factors are likely to be found in careful metallurgical study of the stressed and unstressed areas and detailed attention to the mode of failure, of which there are several in surface fatigue. For example, if the failures have apparently initiated at the depth of orthogonal or maximum shear stresses, then it is doubtful that thermal stresses or localized heating caused by frictional or adiabatic temperature rise have played an important part. On the other hand, I have seen many bearings that failed from the surface, particularly in cases where there is evidence of some sliding which may be caused by retainer resistance.

It does not seem necessary for the shearing stresses produced at the surface to exceed those below the surface for the failure to originate at the surface. There is much to be said for the postulate that surfaces are inherently weak simply because they are surfaces.

Although the authors have found a degree of relative correlation of the four lubricant tests with the full scale bearing, the actual stress level/life relationships from the five ball test and the 7208 size bearings are somewhat in disagreement. I would like

⁸ Staff Engineer, Research Department, Caterpillar Tractor Co., Peoria, Ill.

to know whether the modes of failure were identical in the two types of tests and whether the authors have some explanation for this difference. Here again, a serious study in the direction of explaining the difference may contribute greatly to our general understanding of bearing performance.

W. E. Littmann⁹

The authors have contributed valuable data on the influence of lubricant parameters on the fatigue life in rolling contact. It is most interesting to see that the influence of lubricants upon rolling contact fatigue in the 5 ball rig at 800,000 psi initial maximum Hertz stress compares favorably with results on actual bearings tested at 322,000 to 349,000 psi maximum Hertz stress. What material were the 7208 bearings made from? Was it possible to determine whether fatigue failures in the bearing and 5 ball rig tests initiated at the surface or below the surface? Since both rig and bearing tests reported involve significant tangential forces in the contact area, do the authors expect the same influence of lubricant on fatigue life under conditions of "pure" rolling contact?

In the lubricant film thickness studies, does the application of tangential force to the contact area affect the film thickness? Were any film thickness measurements made under conditions of combined rolling and sliding similar to the rig and bearing test conditions?

Can the authors supply information on the heat-treatment and hardness of the 52,100 disks used in the film thickness measurements? The indicated onset of plastic deformation at 600,000 to 650,000 psi is somewhat higher than we have observed and which has been reported in the literature for low rolling speeds. Was any attempt made to determine whether the distribution of contact stress by the lubricant film shifted the onset of plastic deformation to a higher stress level than for static loading or low rolling speeds?

Authors' Closure

The authors would like to thank the discussers for their comments and interest.

Dr. Burton's hypothesis concerning the increased probability for initiation for fatigue cracks at the surfaces of specimens in rolling contact may be quite plausible when the stresses are very close to Hertzian at conditions of very high loads, very low speeds, and/or with no liquid lubrication. However, even under the stringent conditions reported in the paper, the x-ray measurements indicated definite support and modification of the stresses at the edges of the contact zone, presumably as a result of hydrodynamic oil-film pressures. These lubricant-film pressures, we believe, would eliminate the discontinuity in shear stress at the edge of the contact shown in Fig. 13.

Whether or not differences in the surface pressure induced by elastohydrodynamic lubrication have an appreciable effect on the subsurface stresses would require a more complete analysis before any definite conclusion can be drawn. However, it is possible that, even though the subsurface stresses may not be greatly affected as a result of changes in the contact-pressure profile, the effective stressed volume may be significantly changed. Thus a reanalysis of the Lundberg-Palmgren criterion for the fatigue of rolling contacts may be necessary.

As a further indication of the effect of elastohydrodynamic lubrication on the contact stresses during rolling contact, it was most pleasing to see the deformation profiles in the rolling direction given by Mr. Kannel, Fig. 14. The characteristic thinning of the lubricant film in the trailing part of the contact area is, of course, further proof of the existence of hydrodynamic lubrication of the rolling-contact surfaces and a deviation from Hertzian contact. A similar contact configuration is inferred from data

obtained by Kirk [14].¹⁰ The shape of the deformed contact correlates well with existing elastohydrodynamic theories. In fact, the depression near the center of the contact reflects, in our opinion, the presence of a narrow pressure peak in the lubricant film, which results in a shear-stress concentration in the contact at a point below the surface but much closer to the surface than the maximum subsurface reversing shear stress predicted by normal Hertzian theory.

Mr. Kelley mentions the possible effects of surface tractions and heating in the contact region on the stresses critical to rolling-contact fatigue. It is entirely possible that the tractive forces in rolling-contact systems play an important part in fatigue. Traction will be accentuated by asperity contact between the surfaces, which depends on the microgeometry or surface roughness, as well as the ability of the lubricant film to separate the surfaces. Certainly the ability of some lubricants to form hydrodynamic films thick enough to separate hardened-bearing-steel surfaces under high loads and to plastically deform the contact region was demonstrated in the film thickness measurements reported in the paper. Further, fatigue failures of subsurface origin occurred under the aforesaid loads for both the bearing and five-ball fatigue specimens reported herein. The fatigue spalls and incipient failures in both the bearings and ball specimens were similar as evidenced by metallurgical studies of the spalls and the subsurface region of the running tracks.

The experimental evidence reported herein suggests differences in the deformed contact profile with the different lubricants examined, Figs. 8 and 12. These differences probably result from normal pressure redistribution rather than surface traction. (Surface traction should have only secondary effects on the normal deformation measured.) Further, under normal lubricated conditions, tractive forces should not affect the shear stress amplitude (on which we believe fatigue is dependent) by any significant amount [15]. Also, the spreading of hydrodynamic pressures at the edge of the Hertzian contact very likely has the greater effect on the shear stress amplitude and stressed volume under these normal operating conditions. Therefore, the authors wonder what experimental evidence Mr. Kelley has that shows that the normal stress redistribution through the hydrodynamic film is minor.

Research on the effects of ball spin on rolling-contact fatigue [16] indicated that thermal stresses due to sliding may be of extreme importance in accounting for differences in fatigue life with the same lubricant. However, by using the same experimental techniques, the data presented in this paper indicated no inverse correlation between contact temperature induced by different lubricants and fatigue life.

Mr. Kelley states that the actual stress-life relationships from the five-ball tests and the 7208-size bearing tests are somewhat in disagreement. This can be accounted for by plastic deformation occurring at maximum Hertz stresses beyond 600,000 psi. As the amount of plastic deformation increases, the contact stress is reduced from that calculated and the fatigue life is increased beyond theoretical values. Therefore, it is assumed on the basis of the data presented in this paper that there is no reduction in stress at 600,000 psi and a 10-percent reduction in stress occurring at 800,000 psi because of plastic deformation. If this occurs, the stress-life exponent would be approximately 10.5 instead of 9. At a stress of 800,000 psi, 25 percent less capacity would have resulted had no plastic deformation occurred. This indicates that one must be extremely careful to account for plastic deformation in order to extrapolate life results back to lower stress ranges, i.e., below 600,000 psi.

In regard to the inquiry of Dr. Littmann concerning the bearings, they were made of air-melt 52100 steel. Other pertinent data regarding these bearings are as follows:

- (1) Specifications—ABEC 1
- (2) 14 balls, $\frac{15}{32}$ -in. dia

⁹ Chief Research Metallurgist, The Timken Roller Bearing Co., Canton, Ohio.

¹⁰ Numbers in brackets designate Additional References at end of this closure.

- (3) 40 deg contact angle
- (4) Race conformity: inner race, 51.75 percent; outer race, 53 percent
- (5) Race dia (measured to bottom of groove): inner race, 1.8937 in.; outer race, 2.8411 in.

In reply to Dr. Littmann's question on the effect of sliding on film thickness, we were not able to duplicate in the rolling-disk machine the combined rolling and sliding conditions present in the rig and bearing tests. However, it was determined in previous testing [4] that 3-percent sliding had negligible effect on film thickness. In other independent measurements reported by Crook [17], the film temperature rose markedly with only a slight decrease in film thickness when considerable sliding was superimposed on the rolling motion between two disks. This suggests that the process of oil-film formation may occur at the leading edge of the contact area before enough heating occurs to reduce the effective viscosity of the oil in this region.

The effect that tangential forces impart on a lubricant's role in the contact zone of rolling elements requires further study. Therefore, it is difficult to answer with absolute certainty Dr. Littmann's question whether the influence of lubricant under conditions where tangential forces are present will be the same as that under conditions of "pure" rolling. However, fatigue data obtained in the NASA spin rig at room temperature under conditions of "pure" rolling and in the NASA five-ball fatigue tester at 300 deg F where the aforesaid tractive forces are present ranked the same lubricants run in each rig in nearly the same order [6].

The 52100 disks used for x-ray measurement of the film thickness were treated in liquid nitrogen following the quench and tempered at 450 deg F, with several cooling and tempering cycles to transform all of the retained austenite. The onset of gross plastic deformation during running was determined by detecting deviations of the unloaded disk profile after each loading from the original undeformed shape. However, no attempt was made to determine if the onset of plastic deformation was shifted to a higher stress level when lubricant was present as compared to unlubricated conditions.

The authors in closing wish to thank Mr. H. R. Bear of the NASA Lewis Research Center for his metallurgical analysis of the bearings and ball specimens using the techniques reported in [18].

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